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M.YE. DEYCH, V.S. DANILIN, G.B. TSIKLAURI AND V.K. SHANIN

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RESEARCH ON THE FLOW OF MOIST VAPOR IN AXISYMMETRICAL LAVAL NOZZLES WITHIN A BROAD RANGE OF DEGREES OF MOISTURE

M.Ye. Deych, V.S. Danilin, G.B. Tsiklauri and V.K. Shanin

ABSTRACT: This article contains the experimental results on pressure distribution, the coefficients of discharge and velocity of an axisymmetrical Laval nozzle operating on moist steam and within a wide range of initial degrees of dryness (x_0 from 1 to 0.20). The experiments were conducted on a nozzle profiled by the method of characteristics for $K = 1.3$ with a calculated number $M = 1.8$. Certain uncalculated operating regimes of the nozzle are investigated. The simplest theoretical models of a double-phase flow are examined. The experiments showed a substantial decrease in the coefficient of velocity of the nozzle in the double-phase region.

Of all the known methods for experimental investigation of energy losses in double-phase flows, the most preferred, as the analysis of reference [1] shows, is the method based on the measurement of the reactive force of a stream in a double-phase medium flowing from a conduit. This method is used in our experimental investigation. /327*

A specially developed thermal scheme of the apparatus allowed us to have at the entrance to the investigated nozzle an almost equilibrium double-phase flow with humidity of up to 80% with initial pressure p_0 of up to 4.5 abs. atm which was obtained by mixing saturated steam with water injected into the steam flow through centrifugal sprayers, while the temperature of the water in front of the sprayers was maintained practically equal to the saturation point which corresponds to the initial pressure of the steam. The initial degree of moisture was determined from the equations of the heat and matter balance based on measurements of the temperature of the water and steam, the flow rates of the water (the measuring disk), the drainage and the flow rate of the mixture through the nozzle (the measuring tanks for drainage and condensate).

Figure 1 shows a diagram of the experimental section. The operative section of the apparatus was horizontal and composed of two components: an external immovable one (1) and an internal movable one (2) which acts as a humidifier. The internal component with

* Numbers in the margin indicate pagination in the foreign text.

the nozzle (3) attached to it was connected to Sylphons (4) and (5) through which saturated steam and water are supplied to the collectors (6) of the sprayers (7). To drain the moisture film from the walls of the frame we installed a cut-off valve. A rod (10) transmitting the reactive force of the stream flowing from the nozzle, on an element of the suspension system, was inserted through the back wall of the external component using a Sylphon connector (9).

The weighing unit of the zero shift is single component, tensometric scales composed of a beam (11) which detects the reaction amplification, and a calibrated angular arm (12) with weights M_1 and M_2. With calibration of the circuit the weights imitate the reactive force. The electrical circuit of the scales is a four-component unbalanced bridge (Fig. 2) composed of two operating (O_1 and O_2) and two compensating (C_1 and C_2) transducers of approximately equal resistence. In the experiments we used p-type crystal silicon sensors with a resistence of about 120 ohm, a coefficient of tensor sensitivity of 100-120 and a specific resistance of 0.02 ohm·cm. The parameters of the sensors allowed us to avoid complex amplification systems and to use a direct scheme for transmitting the signals from the sensors to the EPD-32 potentiometers, included in the measuring diagonal of the bridge. For the most precise measurement of the reactive force and the exclusion of the secondary forces, caused by temperature deformations and pressure drop on the /328

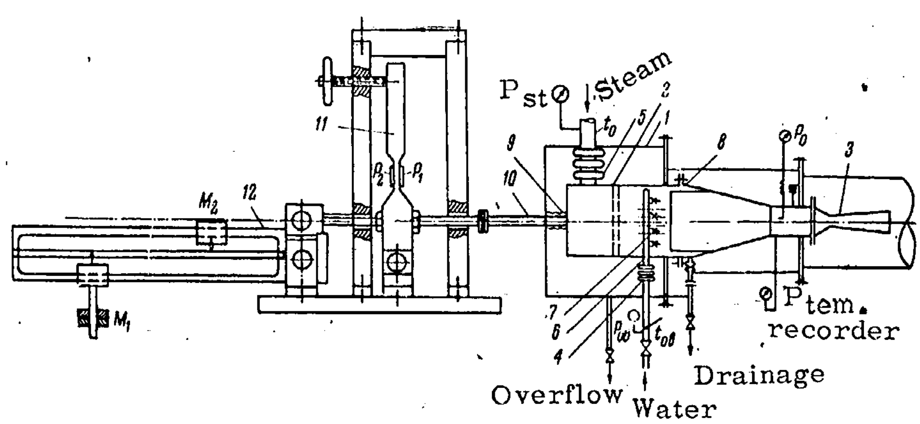

Fig. 1. A Diagram of the Experimental Set-Up.

Sylphon connector (9), we used the following experimental procedure. With a fixed operating regime let the reading of the instrument be n delineations, and after a sudden cut-off in the supply of steam and water and with the previous value of the counterpressure let the reading be n_0 delineations. It is easy to show that in this case all the similar forces with the operating and displaced apparatus are equal and the following equation holds:

$$(n-n_0)\delta=M_{exp}\bar{c}_{exit}+(Pcs-Pa)F_{exit}+M_{exp}\bar{c}_{ent} \quad (1)$$

where δ is the scale value of the instrument, determined by the calibration of the equipment in a "heated" state; M_{exp} is the experimentally measured mass flow; c_{exit} is the velocity of the flow on the cross section of the nozzle averaged according to impulse; p_{es} is the static pressure on the cross section of the nozzle, determined experimentally; p_a is the counterpressure; F_{exit} is the area of the exit cross section; $\bar{c}_{ent}$ is the mean flow rate at the entrance to the nozzle.

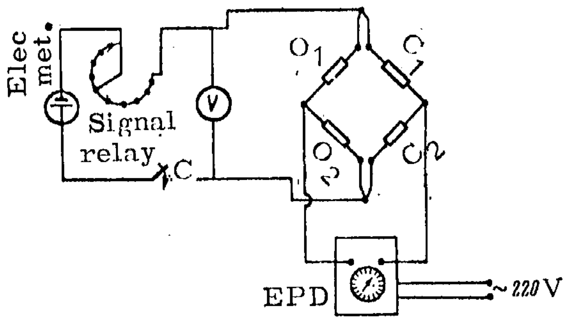

Fig. 2. The Electric Measuring Circuit.

The experimental procedure is explained in detail in a report of the Moscow Power Institute[1], which shows that the maximum error in our determination of the velocity coefficient of the nozzle ϕ_n is 4.5 - 5.0%.

The object of investigation was a supersonic axisymmetric nozzle, computed by the characteristics method for an adiabatic indicator k = 1.3. The convergence section of the nozzle was shaped like a lamnescate. The nozzle was characterized by the following computation parameters: M_p = 1.8; λ_p = 1.59, ε_p = 1.72. To investigate the static pressure distribution along the length of the /329
nozzle there were 22 apertures with a diameter of 0.8 mm on the wall of the nozzle. The diameter of the smallest cross section of the nozzle was 32.55 mm, the diameter of the exit cross section was 40.05 mm and the overall length of the nozzle was 125 mm. The length of the supersonic section was 89 mm.

This nozzle is clearly not optimal for operation on moist steam with a low degree of dryness. However, it was chosen as an initial object of investigation since with this nozzle we accumulated valuable experimental material with its operation on moist steam with a low degree of moisture (up to 20%)[2]. We should also note that at the present time there are no practical recommendations for designing supersonic nozzles to operate on a double-phase medium with a low degree of dryness.

Simplified models of double-phase streams [2] do not reveal the complex mechanism of exchange in a nozzle, yet the numerical results, which were obtained rather easily, provide an opportunity to evaluate certain particular cases. However, if it is possible to compare the experimental data on flow rate and pressure distribution with the results of calculations according to any theoretical

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[1] M.Ye. Deych, G.B. Tsiklauri, V.S. Danilin and V.K. Shanin, Report of the Moscow Power Institute, No. 118/67, 1968.

[2] G.B. Tsirklauri, Cand. Dissertation, Moscow Power Institute, 1964.

model, then it is possible to find the energy characteristics by theoretical means only for a maximally equilibrium process.

Based on the equilibrium model of the flow we made assumptions about the isentropic nature of the flow of mixture with thermal equilibrium of the phases, equality of their velocities and a correlation between the thermodynamic characteristics of an equilibrium double-phase system and the plane interfaces [3]. The main point of the calculation was a determination of the flow rate across the nozzle. We know that for any compressible medium the calculation of the specific flow rate $\bar{q}$ with constant entropy and a fixed p_0 with various differentials on the nozzle, is reduced to the functions $\bar{q}(\varepsilon_a)$. With $\varepsilon_a = 1$ and $\varepsilon_a = 0$, $\bar{q} = 0$ and $\bar{q}$ is at a maximum at a certain ε_{acr}.

Numerical determination of the function $\bar{q}(\varepsilon_a)$ with the subsequent graphic determination of $\bar{q}_{cr}$ and $\varepsilon_{a,cr}$ [2] was made with p_0 = 1.2 abs. atm and $0 \leq x_0 \leq 1$ on a "Minsk-22" computer with a step over p_a of 0.01 abs. atm and a step over x_0 of 0.05. As a result of the calculations we obtained generalized functions for the critical values $\bar{q}_{cr}(x_0)$ and $\varepsilon_{a,cr}(x_0)$, allowing us to determine the pressure distribution in the nozzle and the values of the specific and absolute impulses.

The model of a flow with a fixed composition is based on the following assumptions: (a) the phase velocities are equal; (b) there is no heat or mass exchange between phases; (c) the expansion of steam follows an adiabatic law with a constant k (k = 1.3 and k = 1.135); (d) all the kinetic energy of the flow results from the expansion of the steam; (e) the flow rate and pressure distribution are determined by the geometry of the nozzle and the characteristics of the steam phase.

On these assumptions the basic calculated formulas[3] have the form:

the critical velocity of the mixture is

$$c_{cr,mix}=\sqrt{x_0 \cdot 2p_0 v_{v,st}\frac{k}{k-1}[1-\varepsilon_{cr}^{(k-1)/k}]}=c_{cr,st}\sqrt{x_0}, \quad (2)$$

where $\varepsilon_{cr} = [2/(k + 1)]^{k/(k+1)}$, $c_{cr,st}$ is the critical velocity of steam, $v_{v,st}$ is the specific volume of steam with p_0;

the critical volume of the mixture is /330

$$v_{cr,mix}=x_0[(1/\varepsilon_{cr})^{1/k}v_{v.st}-v_e]+v_1=x_0\Delta v+v_1 \quad (3)$$

[3] M.Ye. Deych, G.V. Tsirklauri, V.S. Danilin and V.K. Shanin, Report of the Moscow Power Institute, No.118/67, 1968.

where v_c is the specific volume of the liquid phase; $\Delta v = v_{cr,st} - v_1$;

the critical flow rate of the mixture through the nozzle is

$$G_{cr,mix} = \frac{c_{cr,st}\sqrt{x_0}}{x_0 \Delta v + v_e} F_{cr} \tag{4}$$

where F_{cr} is the area of the critical cross section of the nozzle.

The fact that $G_{cr,mix} = 0$ with $x_0 = 0$ and $G_{cr,mix}(x_0)$ is at a maximum with $x_0{}^m = v_1/\Delta v$ does not correspond with the true picture of the process and is related to the assumption that the kinetic energy of the flow of mixture is generated only by expansion of the steam phase. As x_0 increases, this assumption becomes more valid. At low steam pressures the value $x_0{}^m$ is very small ($x_0{}^m$ = 0.045% with p_0 = 1.2 abs. atm). It is easy to show that under the given assumptions, the "calculated" operating regime of the nozzle will be the regime corresponding to the calculated value of ε_a with a given k. The results of the calculations according to the fixed composition model are used below for comparison with the experiment.

Figure 3 shows the experimental data on the flow rate characteristics of the examined nozzle. It follows from these graphs that both flow models give lower values for the flow rates, therefore, $\mu = G_{exp}/G_{th}$ is substantially greater than unity. The results of reference [2], obtained with high values of p_0 (up to 70 abs. atm) lie between the theoretical curves. Are the given theoretical models extreme cases? For the equilibrium model, of a double-phase flow, the fact that the actual flow rate is higher than its theoretical value is a rather widely known fact. However, it suggests that under certain conditions the structure of the flow and its heterogeneity, the presence of a low-velocity film of the wall of the nozzle, the initial slip, the absolute dimensions of the conduit and the operating regime of the nozzle can lead to a decrease in the flow rate of the mixture in comparison with the equilibrium. Analogous arguments should also pertain to the fixed composition

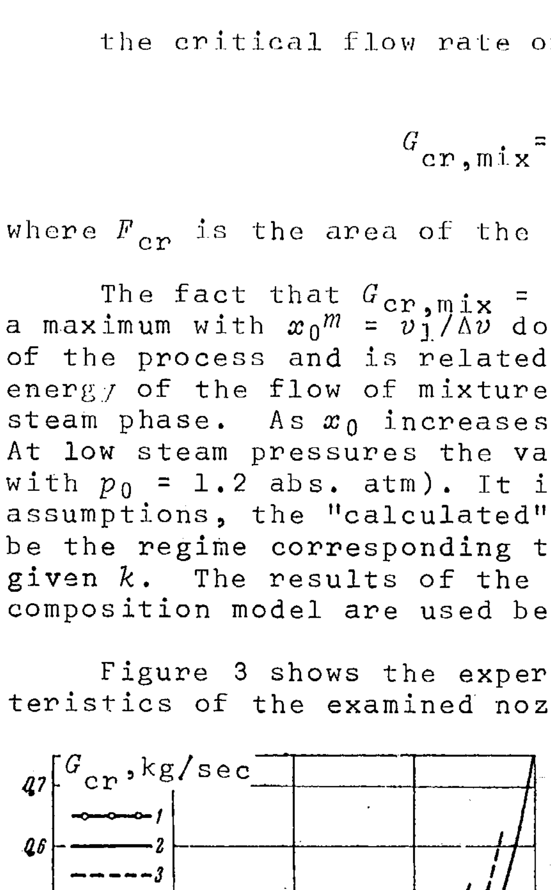

Fig. 3. The Critical Flow Rate and the Coefficients of Moist Steam (p_0 = 1.2 abs. atm; p_a = 0.15 - 0.20 abs. atm): (1) The Experiment; (2) The Equilibrium Model; (3) The Fixed Composition Model.

model which, as follows from its assumptions, does not include all the possible conditions for realizing the maximum flow rate.

Figure 4 shows the experimental curves for the static pressure distribution along the length of the nozzle with p_0 = 1.2 abs. atm and various counterpressures for y_0 = 0.22 and y_0 = 0.70. For purposes of comparison, the graph shows the computed curves for the fixed composition model with k = 1.3 and k = 1.135 and the curves of the equilibrium expansion process. Apparently these experimental curves are typical for two different ranges of initial moisture and are fundamentally different from one another.

With small values of y_0 in the expanding section of the nozzle, a condensation jump occurs converting the flow from a supercooled state to an equilibrium state. Before the condensation jump the experimental curve is rather close to the computed curve with k = 1.3 which corresponds to the data obtained earlier. /331

It follows from Figure 4 that with a counterpressure $\varepsilon_a \leq \varepsilon_{a,calc}$ the static pressure after the condensation jump decreases monotonically, although on the cross section of the nozzle it is significantly higher than the computed pressure with k = 1.3 and k = 1.135. With a certain $\varepsilon_a > \varepsilon_{a.calc}$, after the condensation jump there is a compression shock (shown in the curve for ε_a = 0.53). In this case

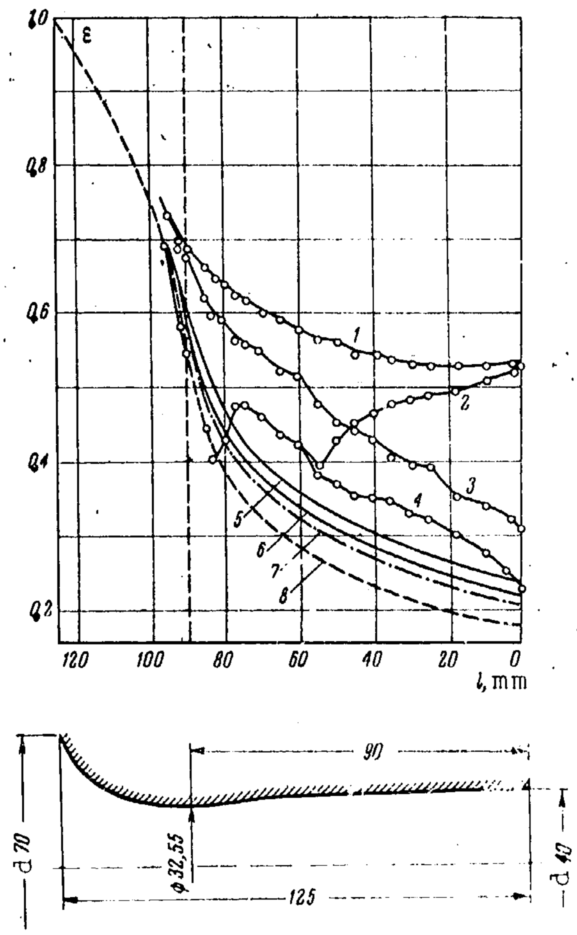

Fig. 4. The Static Pressure Distribution Along the Length of the Nozzle (p_0 = 1.2 abs. atm): (1) y_0 = 0.70; (2) y_0 = 0.22, ε_a = 0.53; (3) y_0 = 0.70; (4) y_0 = 0.23, ε_a = 0.15; (5) y_0 = 0.70; (6) y_0 = 0.22; (7) K = 1.135; (8) K = 1.3; (1-4) The Experiment; (5,6) The Equilibrium Model; (7,8) The Fixed Composition Model.

the pressure on the cross section of the nozzle is equal to the counterpressure. As we know, all these characteristics describe a supersonic flow regime.

With a high initial moisture the pressure distribution has mainly another character. Thus, for y_0 = 0.7 and $\varepsilon_a < \varepsilon_{a,calc}$ the pressure decreases monotonically, while there is a characteristic increase in in pressure at the analogous points of the nozzle as the initial moisture increases. If the flow is supersonic then the disappearance of the condensation jump should be explained by the difficulty of strong supercooling of the steam flow due to the presence of a large amount of liquid. In fact, pressure jumps were not detected in [2].

However, as the experiment showed, in the zone of the highest initial moisture the condensation jumps disappear and the pressure on the cross section of the nozzle continuously follows the counter pressure in the zone $\varepsilon_a > \varepsilon_{a,calc}$ (Fig. 4 shows one of these curves for $\varepsilon_a = 0.53$). Thus, in flows with a high degree of moisture, beginning from a certain $\varepsilon_{a,calc}$, the perturbations spread in the opposite direction from the subsonic flow. This experimental fact does not correspond with the data of reference [2] whose authors detected "ripple" and "diffusion" shock waves in the limits of the nozzle in the regimes of overexpansion.

Figure 5 shows the values of the relative pressure on the cross section of the nozzle and the throat of the nozzle with $\varepsilon_a = 0.15 < \varepsilon_{a,calc}$, denoting the increase in pressure as the initial moisture increases.

It follows from the experimental curves in Figure 6 that the change in the velocity coefficient $\phi_n = c_{exit}/c_{th}$ (c_{th} is the theo- /332

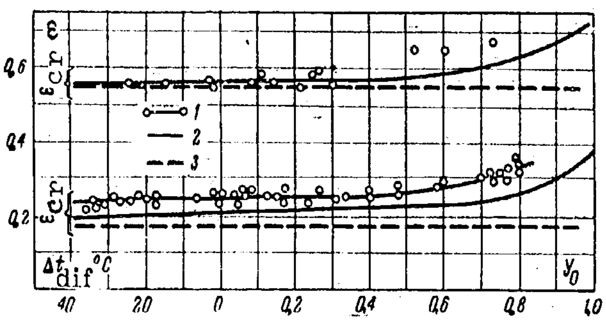

Fig. 5. The Relative Pressure in the Throat and on the Cross Section of the Nozzle as a Function of y_0 (p_0 = 1.2 abs. atm; p_a = 0.15 abs. atm): (1) The Experiment; (2) The Equilibrium Model; (3) The Fixed Composition Model.

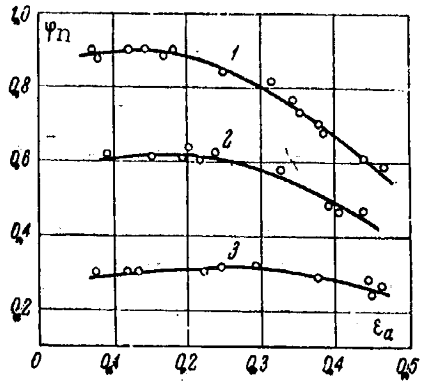

Fig. 6. The Function Relating ϕ_n to the Operating Regime of the Nozzle (p_0 = 1.2 abs. atm): (1) Δt = 20°C; (2) y_0 = 0.33; (3) y_0 = 0.51; (4) y_0 = 0.73.

retical velocity of the equilibrium process of efflux, computed according to the heat differential from p_0 to p_{cr}) as a function of the relative differential on the nozzle ε_a is qualitatively analogous to the behavior of the curves for a single-phase gas medium. In the calculated regime and regimes with $p_a < p_{cr}$ the losses in the nozzle are minimal and vary little with ε_a, and in the regimes with a greater counterpressure there is a sharp decrease in ϕ_n caused by the change in the wave and eddy losses [4]. We know that an increase in pressure in a system of jumps, arising in a supersonic stream with $p_a > p_{cr}$, expands through the subsonic section of the boundary layer inside the nozzle and leads to a redistribution of the parameters in the exit cross section. For double-phase streams these phenomena are even more apparent. There is a particularly strong influence of the low-temperature fluid film on the wall of the nozzle.

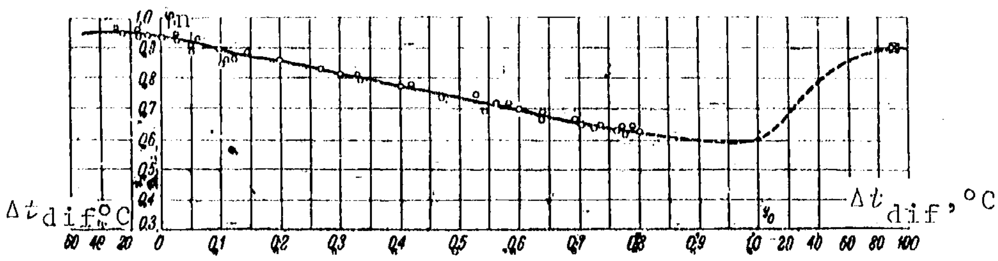

Fig. 7. The Velocity Coefficient of the Nozzle $\phi_n{}^p$ in the Calculated Regimes (p_0 = 1.2 abs. atm).

Figure 7 shows the values of ϕ_n in the computed regime as a function of the initial parameters. Note the fact that ϕ_n decreases sharply as the initial moisture y_0 increases.

Apparently this is a result of the large losses accompanying the mechanical interaction of the steam and liquid phases. Clearly the value of these losses is determined, first of all, by the structure of the double-phase flow.

With an initial drop structure of the stream in a gradient flow, a significant slip must inevitably arise which leads to high
mechanical losses. It is probable that in parts of the supersonic /333
gas flow with certain values of the slip coefficient, the streamlining of the liquid drops is accompanied by significant wave losses. If at the entrance to the nozzle the liquid phase is continuous (bubble structure) and if the structure of the flow does not change within the limits of the nozzle, then we can expect the total absence of slip and high values of the velocity coefficients. Finally, for a perfectly fluid flow on the condition that the flow is uninterrupted, we can expect the same values of ϕ_n as for a gas flow at low velocities (an incompressible liquid), which according to the available data are 0.88 - 0.92. This is the hypothetical form of the function $\phi_n(y_0)$ in the zone of high degrees of moisture, in the vicinity of y_0 = 1 and in the zone of water underheated to saturation (underheating to 100° C is shown in Fig. 7 by the broken line).

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